Oil Transport Mechanisms Inside Semi-Hermetic Reciprocating Compressors For CO2 Applications

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ABSTRACT

In addition to the gaseous refrigerant an oil-lubricated reciprocating compressor also transports a certain amount of oil into the refrigeration circuit. The amount of oil carried out of the compressor is dependent on the operating conditions and the internal compressor design but is usually assumed to be in the range of a few mass percent. With respect to installation costs, reliability and efficiency of the whole refrigeration system the oil circulation ratio (OCR) should be kept as low as possible.

In order to get a better understanding on how oil is transported inside a reciprocating compressor for CO_2 applications, the internal flow pattern was analyzed and possible oil transport mechanisms were investigated. The compressor was therefore divided into different subsections. Based on these subsections experimental studies on the potential of each individual oil transport mechanism were carried out.

In order to measure the OCR at different operating conditions a new test rig for CO_2 compressors was built up. The internal design of the tested compressor as well as the test setup were gradually modified to allow the separate investigation of the different subsections and thus the potential of the oil transport mechanisms on reducing the overall OCR.

The measurements clearly identified the oil transport mechanisms and gave indications for further reduction of OCR in reciprocating compressors.

1. INTRODUCTION

In positive displacement compressors oil is used for lubrication of the moving parts and to assist the sealing of the compression chambers. By the refrigerant flow inside the compressor a part of this oil is entrained and carried out of the compressor. While the oil transport in refrigeration systems, its dependencies on the operating conditions and the influence of oil on heat transfer and system efficiency is well investigated, there are nearly no published studies on the oil transport mechanisms inside reciprocating compressors for CO_2 applications.

The oil retention in a refrigeration system is depending on the oil circulation ratio (Lee, 2003). Too high OCR can lead to an insufficient amount of oil inside the compressor and hence reduces the reliability. In addition, a too high amount of oil in the system decreases the heat transfer coefficient of the heat exchangers (FKT, 2008) and increases the pressure drop (Hwang *et al.*, 2004). In consequence, an excessive OCR decreases the efficiency of the refrigeration system. In order to suppress this negative effects in common refrigeration systems oil separators in the discharge line are used but increase the installation costs of the system. Because the OCR can differ in the several sections of the system it is reasonable to define an additional quantity in order to characterize the oil amount being carried out of the compressor. For this reasons the oil discharge ratio (ODR) is used in the following description. The ODR is defined as the oil mass flow related to the sum of the oil and refrigerant mass flow discharged by the compressor and should be kept as low as possible.

$$ODR = \frac{\dot{m}_{oil}}{\dot{m}_{oil} + \dot{m}_{ref}} \tag{1}$$

With respect to further reduction of ODR in semi-hermetic reciprocating compressors it is necessary to understand how the oil is transported by the refrigerant inside the compressor. Therefore, a theoretical study on the internal flow pattern and the possible oil transport mechanisms is presented in this paper. Furthermore, the experimental investigation of the mechanisms is shown and discussed.

2. INTERNAL DESIGN OF COMPRESSOR AND INTEGRATION INTO REFRIGERATION SYSTEM

The investigated semi-hermetic reciprocating compressor is designed for transcritical CO_2 applications and has a displacement of 11.6 m³/h at 1750 rpm (60 Hz). Figure 1 shows a cross section of the compressor. It has to be mentioned that the internal design of the compressor is representative for such type of compressor. The compressor housing is internally divided by a solid wall into two major sections, the motor case and the crankcase. The electric motor which drives the crankshaft of the compressor is located in the motor case. Inside the crankcase the crankshaft, connecting rods and pistons are located as well as the oil supply. The oil supply consists of a rotating disc, mounted on the crankshaft, which transports oil out of the oil sump into the so called oil pocket. This oil pocket allows the permanent oil supply through the crankshaft towards the bearings.

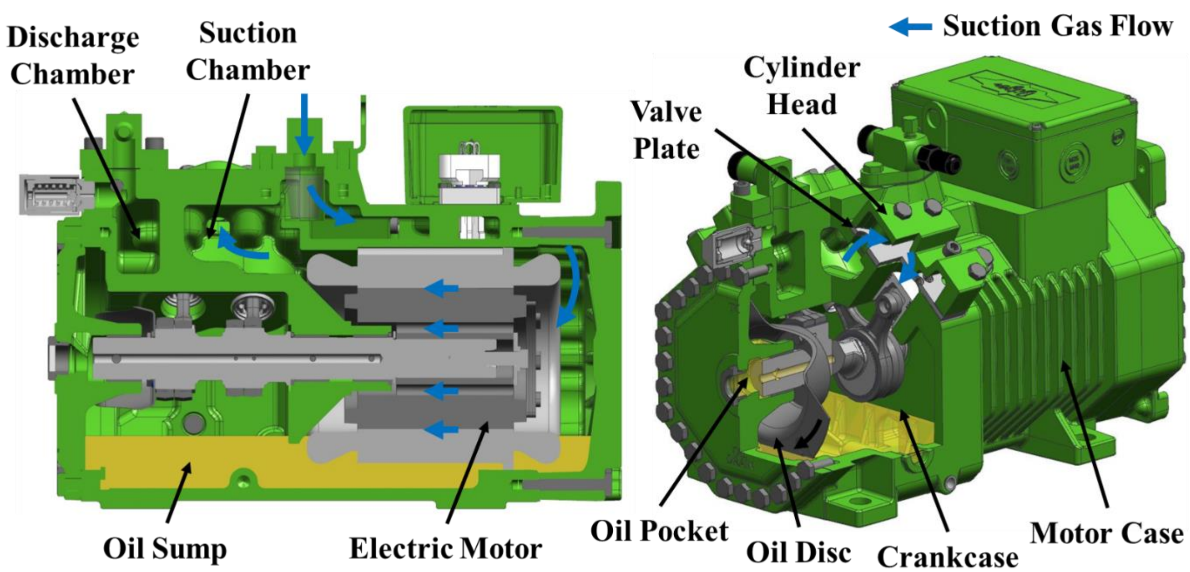

Figure 1: Structure of compressor and suction gas flow

The suction gas flows into the compressor and through ducts towards the electric motor. In order to remove the motor waste heat comparably cold suction gas flows through tiny channels inside the motor. Behind the motor the suction gas flows into the suction chamber and through the suction valves into the cylinders. The gas is compressed and pumped across the discharge valves into the discharge chamber towards the high side of the refrigeration system.

With respect to the oil transport inside a semi-hermetic reciprocating compressor it is necessary to distinguish between two different approaches in order to handle the oil management in common refrigeration systems. In systems with oil separator the amount of oil that is returned to the compressor by the suction gas can be assumed to be comparably low. The ODR is therefore only depending on the amount of oil entrained by the refrigerant flow inside the compressor. In systems without oil separator the amount of oil carried out of the compressor and returned through the suction side needs to be equal under steady-state conditions. Hence, the ODR is depending on both − the oil amount entrained internally as well as the oil returned by the suction gas because this oil has to be separated inside the compressor. Previous internal investigations have shown that both system approaches result in very different oil transport characteristics. This is why this paper is only focused on systems with oil separator.

Because of the comparably low vapor pressure of the oil the calculated maximum amount of oil transported in vapor state on the suction side is less than 1 ppm. Compared to typical ODR values in the range of a few mass percent the oil in vapor state is negligible and the transported oil is assumed to be only liquid.

3. THEORETICAL DISCUSSION OF INTERNAL OIL TRANSPORT MECHANISMS

3.1 Oil flow out of the main bearing

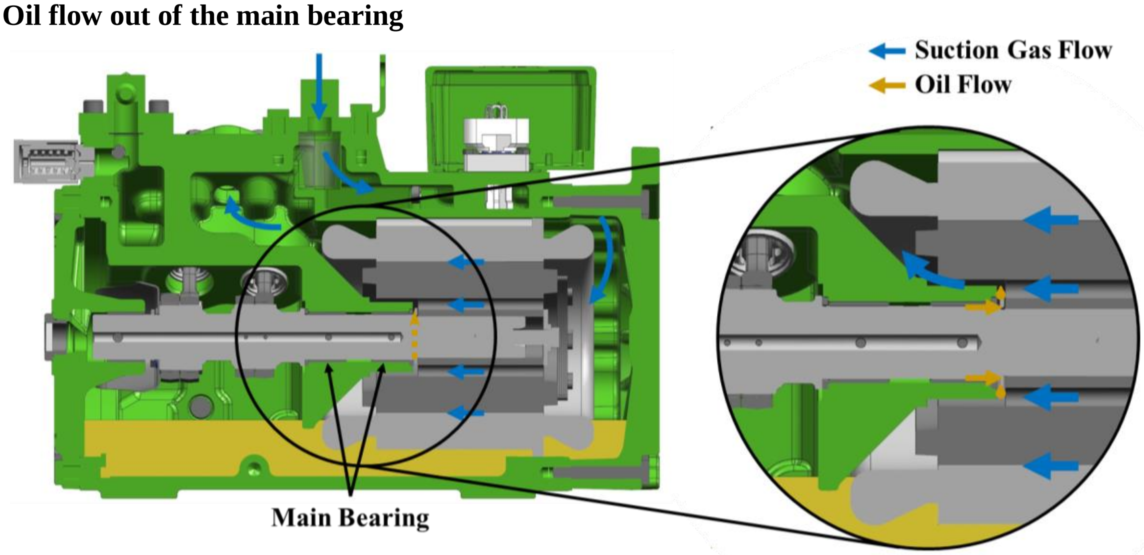

Figure 2: Mechanism oil flow out of the bearing

The first potential entrainment of oil takes place close to the main bearing of the compressor (figure 2). The suction gas passes near the lubricated main bearing. Oil flowing out of the bearing can be entrained by the suction gas flow.

3.2 Refrigerant flow inside the crankcase

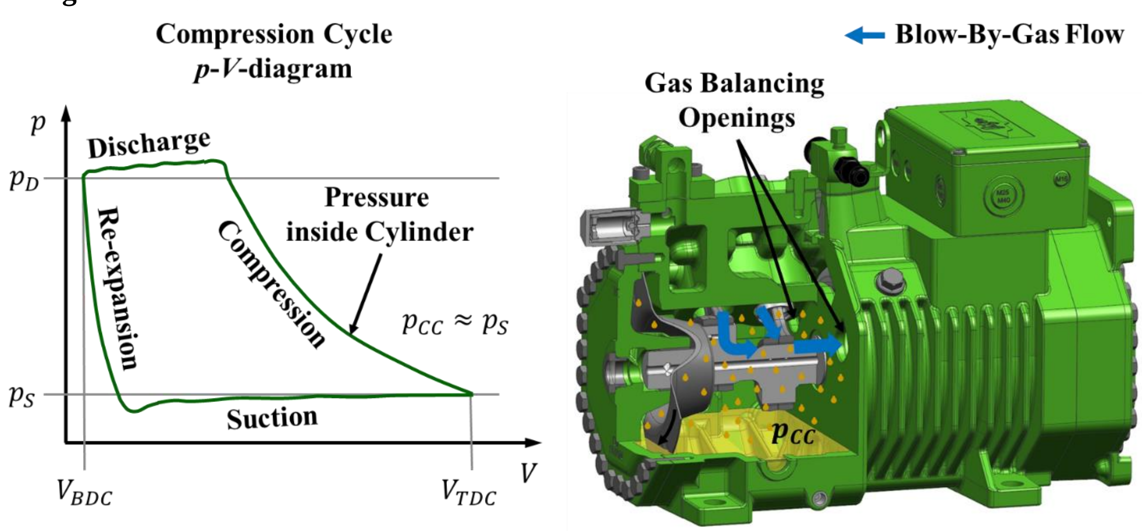

Figure 3: *p*-*V*-diagram and mechanism blow-by-gas

Caused by the lubrication of the bearings and the rotation of the crankshaft small oil droplets are generated inside the crankcase. These droplets can be entrained by the refrigerant flowing through the crankcase. However, during the compression, discharge and re-expansion phase of the compression cycle the pressure inside the cylinder is intermittently higher than the crankcase pressure p_{CC} (p-V-diagram, figure 3). This cyclic pressure difference between the cylinder and the crankcase leads to a leakage at the piston, piston rings and the cylinder wall and refrigerant flows back into the crankcase (blow-by-gas). In order to ensure a gas balance between the crankcase and the motor case this causes a certain gas flow through openings in the wall between the two internal chambers. Hence, the blow-by-gas flows through the crankcase and transports oil droplets out of the crankcase (figure 3, right). The investigated compressor has two gas balancing openings.

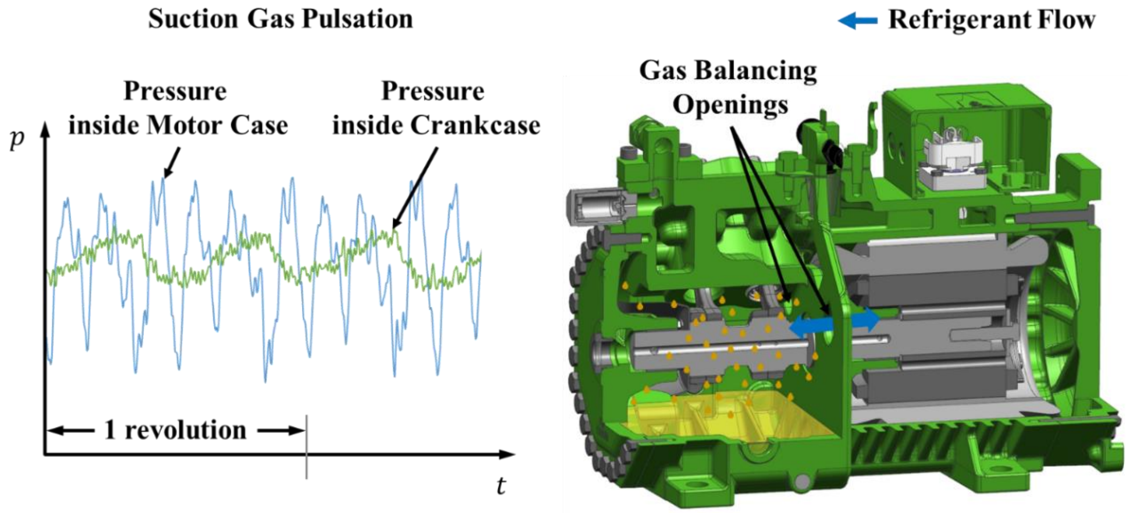

Figure 4: Mechanism suction gas pulsation

The suction gas flow of a reciprocating compressor is always discontinuous which leads to pressure pulsations also in the suction chamber of the compressor and in the motor case. Because of the gas balancing openings a cyclic refrigerant flow into and out of the crankcase takes place (figure 4). This cyclic refrigerant flow is able to transport oil droplets out of crankcase as well.

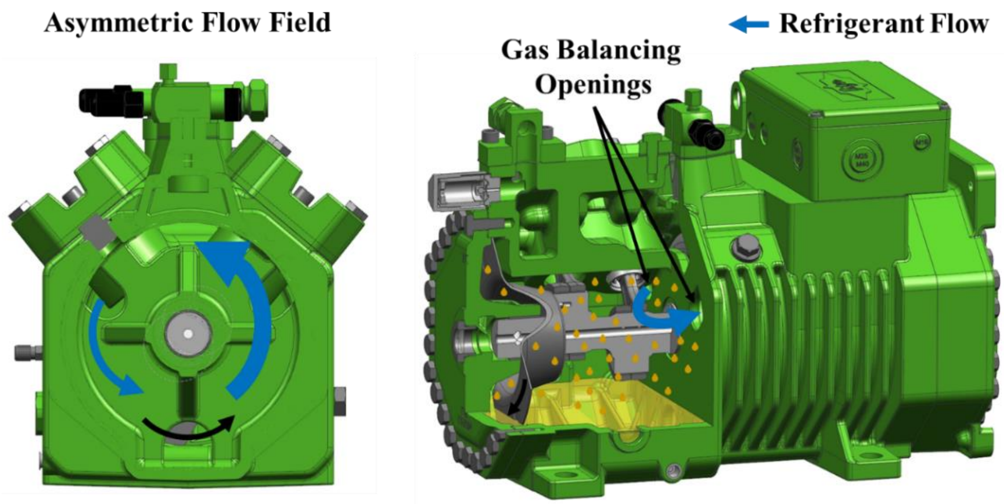

Figure 5: Mechanism secondary flow

As mentioned, the suction gas passes the motor in order to cool it. The rotation of the rotor leads to an asymmetric flow field across the two gas balancing openings and hence induces a secondary refrigerant flow from one opening to the other (figure 5). This additional refrigerant flow through the crankcase is able to entrain oil droplets as well.

3.3 Oil equalization between motor case and crankcase

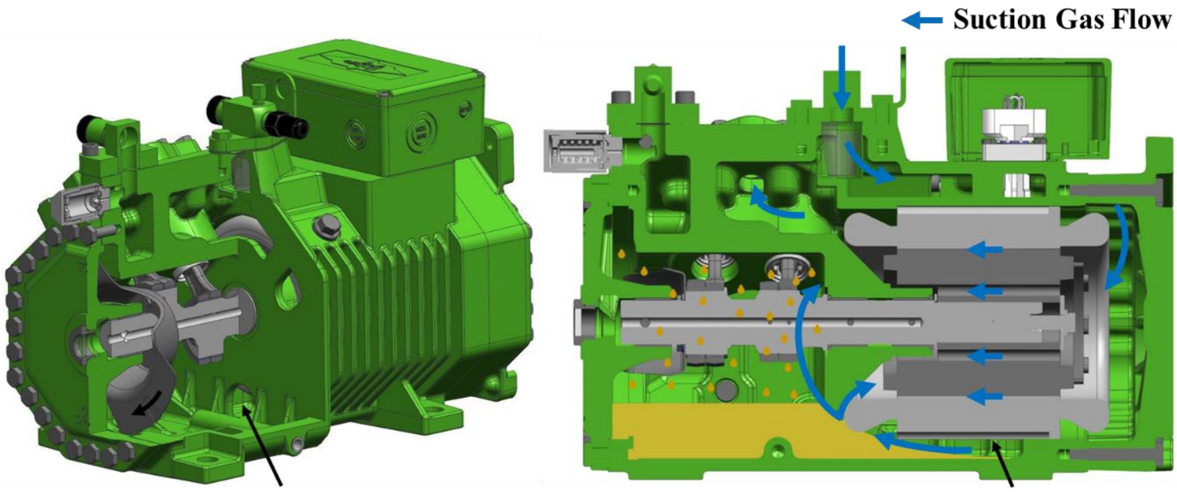

Figure 6: Mechanism oil equalization

In case the compressor is used in a refrigeration system without oil separator the oil returned by the suction gas has to be separated inside the motor case. In order to transport this oil from the motor case into the crankcase an oil return opening in the wall between the two chambers is necessary (figure 6, left). In addition, a specially shaped channel is placed underneath the motor in order to transport separated oil along the motor. Due to the pressure drop of the suction gas flow the oil inside the channel is pushed towards the oil return opening. Hence, a certain amount of refrigerant flows through the channel as well and oil can be entrained near this opening. In addition, a part of this gas flow can enter the crankcase and thus carry oil droplets out of the crankcase (figure 6, right).

3.4 Oil balance inside the cylinder

During the suction phase of the compression cycle the pressure inside the cylinder is lower than in the crankcase (*p*-*V*-diagram, figure 3). The pressure difference can lead to a small leakage flow out of the crankcase into the cylinder, thus in the opposite direction as the blow-by-gas discussed in section 3.2. This can lead to migration of oil into the cylinder. In addition, oil which is not striped off during the downwards movement of the piston might be carried out to the discharge side by the pumped refrigerant flow. The investigated compressor has three compression rings on each piston and no oil scraper rings. With the help of the experimental investigation described later in this paper the potential of this backward type of oil transport is discussed.

4. TEST RIG AND TEST SETUPS

4.1 Test rig

In order to run the compressor at different operating conditions a test rig for sub- and transcritical CO_2 operation was built up. The working principle of the test rig is based on a gas cycle with a two-stage expansion (figure 7). After the compression the refrigerant is expanded to intermediate pressure. Afterwards the refrigerant is cooled down by a water-cooled gas cooler. In the second expansion stage the refrigerant is expanded to suction pressure and returned to the compressor.

The ODR measurement is done by the separation method. Therefore, a high-efficiency coalescence oil separator was installed in the test rig. The separated oil is continuously returned to the crankcase of the compressor. The volume flow of the returned oil is varied as long as the oil level inside the oil separator is steady-state. The mass flow of the refrigerant and the volume flow of the oil are measured as well as the temperature and the pressure of the returned oil. The mass fraction of the dissolved CO_2 in the oil and the mixture density are calculated in order to determine the ODR. Furthermore, the calculated oil mass flow is reduced by the dissolved CO_2 what is added to the refrigerant mass flow.

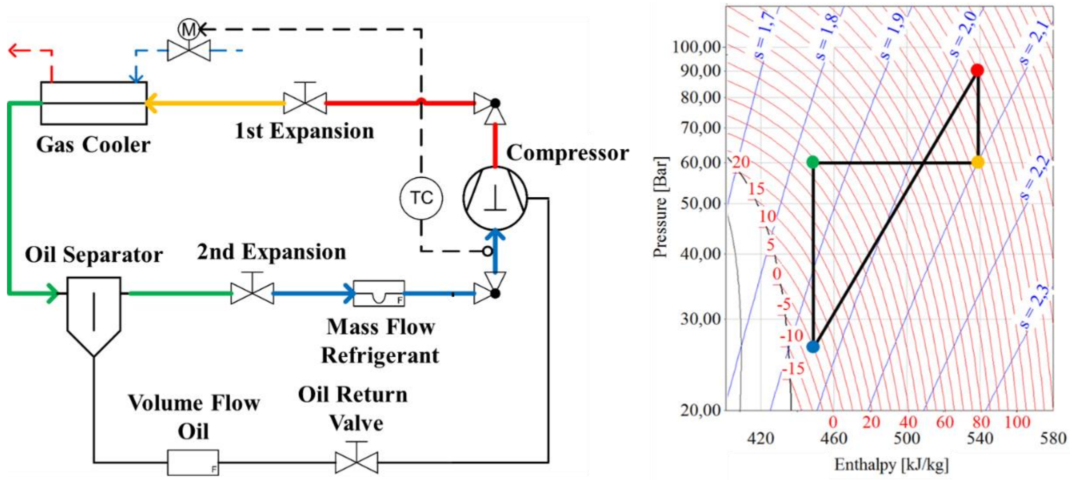

Figure 7: Principle of test rig

4.2 Test setups

In order to investigate separately the influence of the different oil transport mechanisms on the ODR the housing of the compressor was modified and tested in several steps.

Case 1: The oil return and both gas balancing openings in the wall between the crankcase and the motor case were closed. The blow-by-gas was vented out of the crankcase through a second oil separator and afterwards fed back into the suction chamber of the compressor. It was assumed that due to this arrangement the blow-by-gas does not carry any oil into the suction chamber. Furthermore, there is no pressure difference any more between the crankcase and the motor case that would influence the oil flow out of the bearing. In consequence, only the oil flowing out of the bearing can be entrained by the suction gas flow (figure 8). In case not all of this oil is entrained and transported by the suction gas flow an oil pump was connected to the motor case. The accumulated oil in the motor case $\dot{m}_{oil,mc}$ and the separated oil in the second oil separator $\dot{m}_{oil,bbg}$ are pumped back into the crankcase in order to keep the oil level inside the compressor stable.

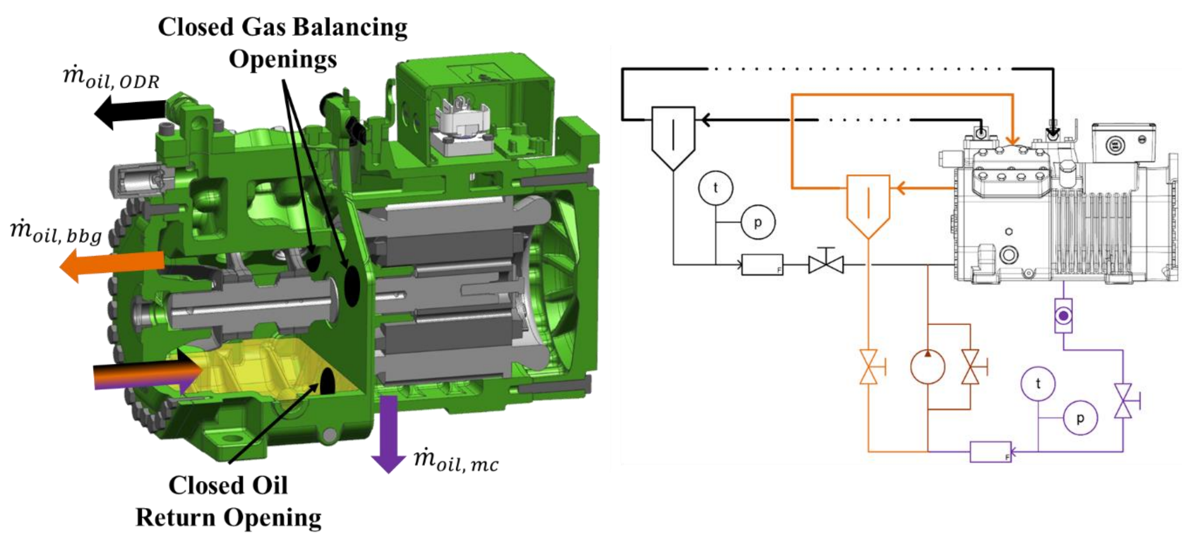

Figure 8: Test setup case 1

Case 2: For the investigation of the refrigerant flow inside the crankcase the gas balancing openings were opened again and the bearing was sealed at the motor side by a shaft seal. The connection between the crankcase and the second oil separator was closed. Accumulated oil in the motor case is again pumped back into the crankcase.

Case 3: In order to isolate the influence of the blow-by-gas and the suction gas pulsation from the secondary flow a test with only one opened gas balancing opening was conducted as well. A further isolation of the influence of the blow-by-gas from the suction gas pulsation is not possible. The further test setup was equal to case 2.

Case 4: In order to investigate the influence of the oil equalization on the ODR the oil return opening was opened again. Furthermore, the bearing was still sealed. In order to ensure that there is a realistic pressure balancing between the crankcase and the motor case, one of the gas balancing openings was also open. Hence, the influence of the blow-by-gas and the suction gas pulsation on the ODR has to be subtracted.

Case 5: All openings were opened and the shaft seal at the bearing was removed again. This allows a test of all the working oil transport mechanisms together as well as the investigation of the overall ODR of the compressor.

Table 1: Overview of test cases

Test Case	Working Oil Transport Mechanisms					Modifications Housing
	Oil Flow out of the Bearing	Blow-By-Gas	Suction Gas Pulsation	Secondary Flow	Oil Equalization	
1	X					Oil return and both gas balancing openings closed
2		X	X	X		Oil return opening closed; bearing sealed
3		X	X			Oil return and 1 gas balancing opening closed; bearing sealed
4		X	X		X	1 gas balancing opening closed; bearing sealed
5	X	X	X	X	X	None

For the investigation of the oil balance inside the cylinders the test setup for case 1 was used. Therefore, the bearing was sealed additionally. In order to test different amounts of oil being transported into the cylinder, oil of the sump was injected into the suction chamber of the compressor. The injected oil flow and the ODR of the compressor were measured in order to determine the oil balance of the cylinder. Furthermore, the dissolved CO_2 in the injected oil was considered as well. In case not all of the injected oil is entrained by the refrigerant flow, the oil pump was used again for pumping the accumulated oil back into the crankcase.

During all tests a specifically formulated POE oil (85 mm²/s at 40°C) for CO_2 applications was used. In order to investigate the influence of the operating conditions on the oil transport mechanisms and the ODR, the parameters suction pressure, discharge pressure, rotational speed (inverter frequency) and oil level of the oil sump were varied within the application envelop of the compressor. The suction superheat in all operating points was 10 K and the rotational direction was clockwise (view on crankshaft, figure 6).

5. EXPERIMANTAL RESULTS AND DISCUSSION

During the tests of case 1 nearly no ODR was measured. It is assumed that the amount of oil flowing out of the main bearing and into the motor case is very small and hence this mechanism is neglected in the following results. However, by using a compressor version with oil pump instead of the design with oil disc for lubrication this effect might be different and should be investigated in further studies. In addition, it was found that the discharge pressure has only a small influence on the overall ODR of the compressor (case 5) and hence won't be discussed further. The main influence on the ODR was measured by varying the suction pressure p_S, the inverter frequency F (rotational speed) and the oil level in the sight glass L_{SG}. All three parameters show an increasing overall ODR by rising the value of the parameter (figure 9 and 10). Furthermore, the ODR caused by the blow-by-gas and the suction gas pulsation (case 3) is comparably small. However, the major potential for further reduction of the overall ODR can be seen in the suppression of the mechanisms secondary flow (part of case 2) and oil equalization (part of case 4).

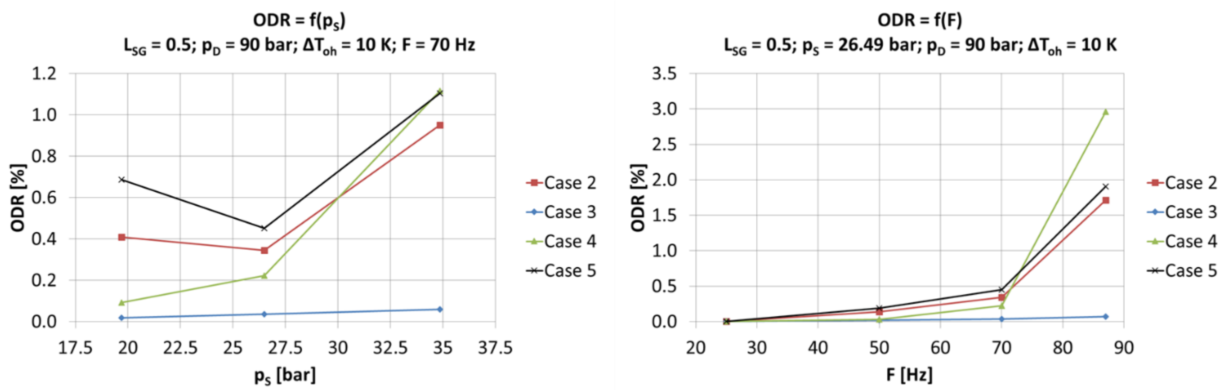

Figure 9: ODR of test cases as function of suction pressure and inverter frequency

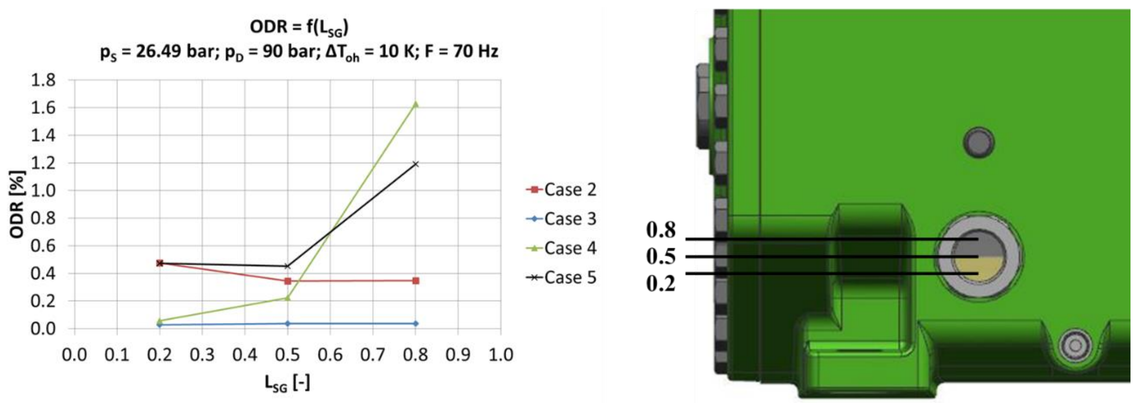

Figure 10: ODR of test cases as function of oil level

Suction pressure: With rising the suction pressure the ODR caused by the refrigerant flow through the crankcase (case 2 and 3) increases. It is assumed that the suction gas pulsation and the volume of the secondary flow increase with rising the suction pressure. This leads to a higher refrigerant volume flow through the crankcase and hence a higher ODR. In addition, the ability of entraining and transporting oil by the refrigerant flow is depending on the density difference between the refrigerant and the oil and hence increases with rising suction pressure. The ODR generated at the oil return opening (case 4) increases with rising the suction pressure as well. The pressure drop of the suction gas along the motor is in a first assumption linearly to the density of the suction gas. This will lead to an increase of the suction gas volume flow through the oil return channel and hence a higher ODR.

Inverter Frequency (rotational speed): The ODR of all test cases increase with rising the inverter frequency. The mass fraction of oil droplets inside the crankcase is found to be depending on the rotational speed. A higher mass fraction leads to an increasing ODR caused by the refrigerant flow through the crankcase (case 2 and 3). Furthermore, it is plausible and confirms the theory that the volume of the secondary flow increases with rising the rotational speed of the rotor. In addition, the pressure drop along the motor is in a first assumption linearly to the square of the volume flow. Hence, the suction gas volume flow through the oil return channel as well as the ODR increase with rising the rotational speed.

Oil level: The ODR caused by the blow-by-gas, the suction gas pulsation and the secondary flow is nearly independent from the oil level. It is plausible that the oil level doesn't have an influence on the refrigerant flow through the crankcase. Furthermore, it seems that the mass fraction of oil droplets inside the crankcase is not depending on the oil level. On the other hand, the ODR of case 4 increases disproportionately with rising the oil level. It is reasonable that the entrainment of oil out of the sump in the area of the oil return opening is depending on the oil level itself.

By comparing the ODR of the several mechanisms with the overall ODR of the compressor it can be concluded that the sum of all mechanisms is not equal to the overall ODR. At some operating conditions the ODR of case 4 is even higher than the overall ODR. This means that interactions between the different oil transport mechanisms exist. For further reduction of the overall ODR the interactions have to be considered and therefore additional investigations are needed. For example, if only the secondary flow is suppressed the overall ODR would increase at the maximum rotational speed or a high oil level (ODR case 4 and 5).

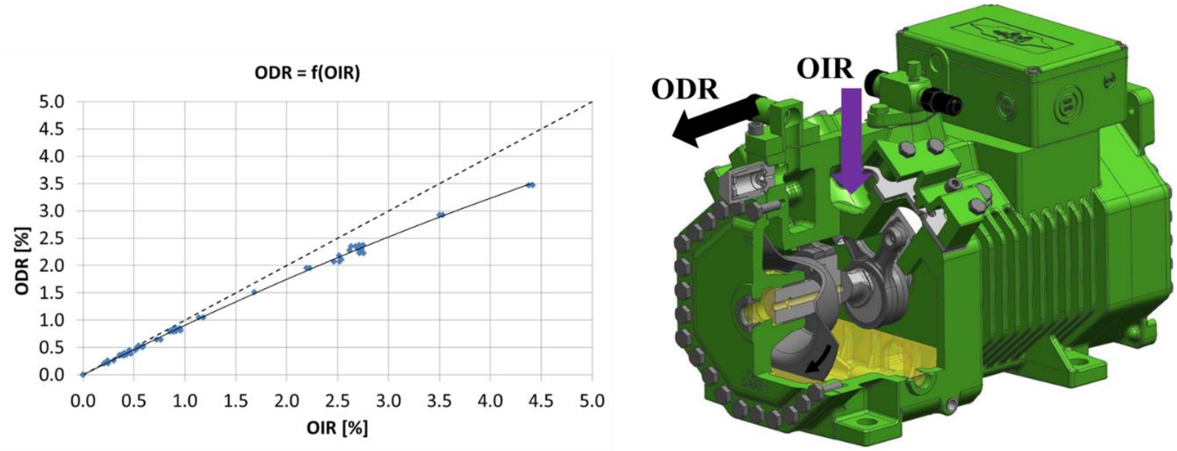

Figure 11: Oil balance inside cylinder

Figure 11 shows the results of the experimental study on the oil balance inside the cylinder. For this purpose the ODR values at all tested operating conditions are plotted over the oil injection ratio (OIR). The OIR is calculated with the mass flow of the injected oil into the suction chamber (reduced by the amount of dissolved CO_2), related to the mass flow of the refrigerant and of the injected oil (inclusive dissolved CO_2). During tests where no oil was injected nearly no ODR could be measured. Hence, a migration of oil during the suction phase of the compression cycle can be neglected. It can even be seen that there is a tendency of displacing oil out of the cylinder into the crankcase. It can be assumed that the blow-by-gas along the cylinder walls is transporting oil out of the cylinder. In addition, it could be observed that the relation between ODR and OIR is nearly independent from the operating conditions.

6. CONCLUSION

In order to understand how oil is transported inside a semi-hermetic reciprocating compressor for CO_2 applications theoretical and experimental studies on the internal oil transport mechanisms were carried out. In principle, there are four possible sections where oil can be entrained by the refrigerant flow – at the main bearing, inside the crankcase, at the oil return opening (motor case to crankcase) and inside the cylinder. Inside the crankcase the blow-by-gas, the refrigerant flow caused by the suction gas pulsation and the secondary flow caused by the rotation of the rotor are able to transport oil droplets out of the crankcase. The following findings could be obtained by the experimental results.

- Overall ODR increases with rising suction pressure, rotational speed and oil level
- ODR caused by oil flowing out of the bearing is negligible (lubrication with oil disc)
- ODR caused by blow-by-gas and suction gas pulsation is comparably small
- Mechanisms secondary flow and oil equalization show major potential for reduction of overall ODR
- Oil migration from crankcase into cylinder is negligible; inside of cylinder has tendency of oil separation

It can be concluded that the oil transport in such a compressor is a very complex topic. Even with these very deep investigations not all effects could be explained completely – especially because of interactions between oil transport mechanisms. Even though no comparisons with other compressors were carried out, the main findings of this paper might be transferred also to other semi-hermetic reciprocating compressor of similar design.

NOMENCLATURE

F	inverter frequency	(Hz)
L_{SG}	oil level in sight glass	(-)
$\dot{m}$	mass flow	(g/s)
OCR	oil circulation ratio	(%)
ODR	oil discharge ratio	(%)
OIR	oil injection ratio	(%)
p	pressure	(bar)
t	time	(s)
ΔT_{oh}	suction super heat	(K)
V	Volume	(m^3)

Subscript

bbg	blow-by-gas
BDC	bottom dead center
cc	crankcase
D	discharge
mc	motor case
oil	oil
ref	refrigerant
S	suction
TDC	top dead center

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